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Infinitely Variable Capacity Control

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INTRODUCTION

This paper discusses the design and development of an infinitely variable capacity control system for reciprocating compressors. Unlike conventional step type systems which vary capacity in fixed increments, this system has an infinite number of steps and is thus stepless. The capacity is controlled to exactly match the demand for gas. This stepless system utilizes hydraulic pressure to hold the inlet valves open during part of the normal compression stroke. Some of the gas drawn into the cylinder is discharged back into the inlet passage. The point at which hydraulic pressure is released and compression allowed to start can be varied to result in a capacity anywhere between zero and 100%. A hydraulic distributor, timed to the crankshaft, applies pressure to inlet valve unloaders after the valves have normally opened. A control device senses discharge pressure or flow, and through a positioner, retards or advances the point at which hydraulic pressure is released from the unloaders. The volumetric efficiency of the compressor is thus adjusted to exactly match the demand.

CONVENTIONAL SYSTEMS

Various capacity control systems are currently in use on large, process type compressors. One of the most common methods is inlet valve unloading. A manually or automatically operated plunger depresses the inlet valve channels, or plates and results in the discharge of gas back into the inlet passage for the entire compression stroke. A variation to this method, but with the same result, is to open or close a centrally located hole in the valve body thus essentially bypassing the inlet valve. Fig. 1 illustrates the results of this type of unloading on capacity and horsepower.

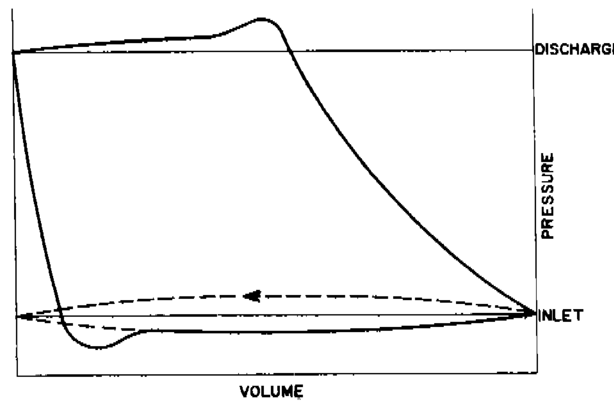


FIG. I P-V CARD SHOWING INLET VALVE UNLOADING

A second commonly used method is to vary the clearance volume of a cylinder in fixed increments. Fixed volume pockets are allowed to communicate with the cylinder bore through a port which can be opened or closed. The volumetric efficiency and horsepower vary as shown in Fig. 2.

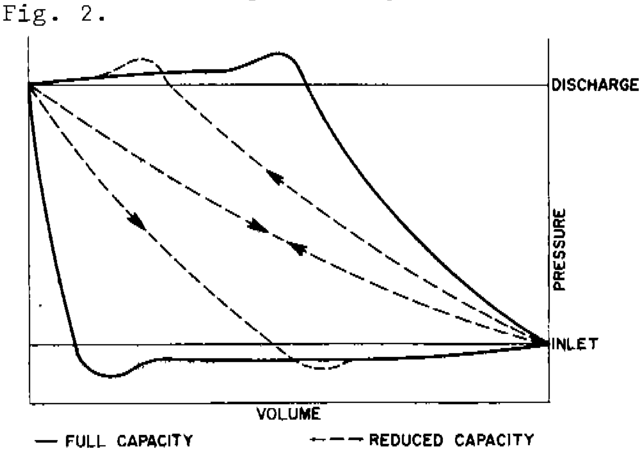


FIG.2 P-V CARD SHOWING CLEARANCE POCKET UNLOADING

A combination of these two methods is frequently used on large double-acting cylinders to increase the number of increments, or steps, and to minimize the percentage change per step. In any of these step type systems, the capacity of the unit at any given time will be greater than or less than the demand. The frequency of operation of a step is a function of the capacity per step, demand, storage volume, and desired closeness of control.

A stepless type automatic clearance control used in recent years has met with limited success. A secondary cylinder and clearance piston is mounted adjacent to the main compressor cylinder, usually in the outer head. The clearance piston can be moved inward or outward progressively changing the clearance volume until the capacity of the unit exactly matches the demand. This method is expensive and from a practical standpoint, can only be applied to the outer end of double-acting cylinders. Infinitely variable capacity control can also be accomplished by throttling the gas at the intake of a compressor or by bypassing the gas from discharge back to intake. Both of these methods waste horsepower and can result in excessively high temperatures unless special precautions are taken. They are rarely used on large process type compressors.

The speed of engine or turbine driven units can be changed to vary capacity. When minimum speed is reached, a further reduction in capacity is accomplished by one of the above methods.

VARIABLE INTAKE VALVE UNLOADING

Delaying the closure of inlet valves to reduce capacity and horsepower results in an indicator card as shown in Fig. 3.

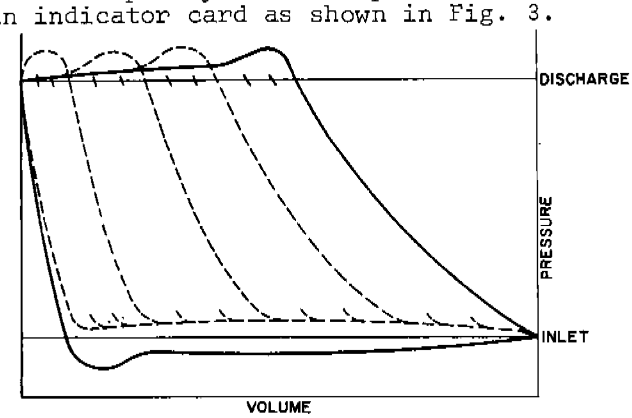


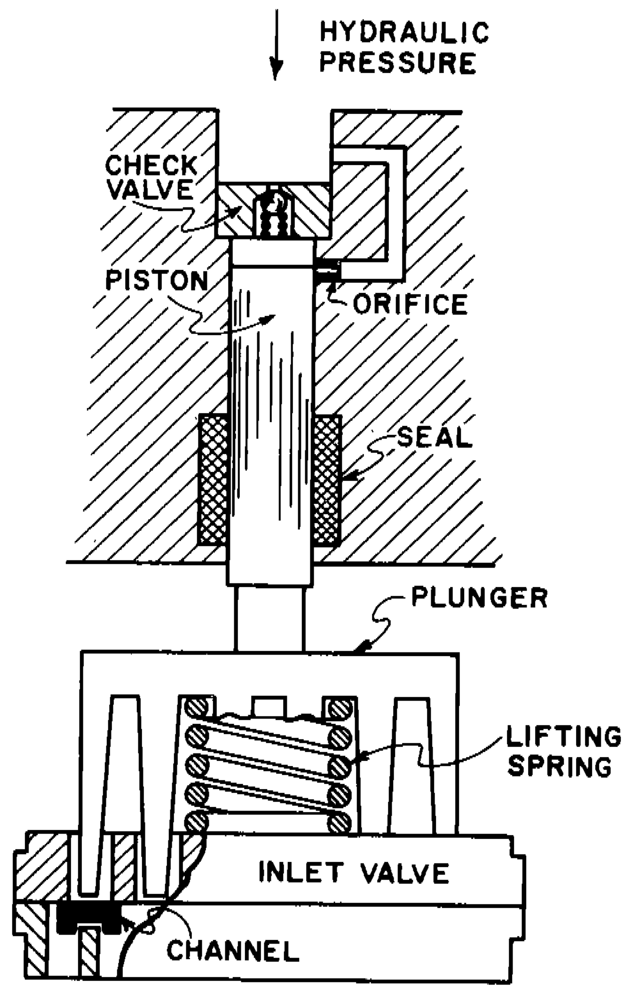
FIG.3 P-V CARD OF INFINITELY VARIABLE CAPACITY CONTROL

One such system uses pneumatic pressure on an external operating piston or diaphragm to preload the valve plates tending to hold them open. During the compression stroke, the pressure drop back through the valve will increase as piston velocity increases. When this force exceeds the preload, the valve will close and compression will start. The maximum backflow force occurs at midstroke when piston velocity is maximum. The volumetric efficiency can thus only be infinitely reduced to about 50%.

The purpose of this paper is to describe the development of a hydraulic system which will vary capacity from 0 to 100%

DEVELOPMENT OF UNLOADER

One basic problem inherent to this method of unloading is inlet valve durability. Since the valve is released while gas is being pushed back through it, it can close at high velocity and be smashed. To eliminate this type of failure, a method had to be devised to control the rate of closing and prevent destructive impact forces. Fig. 4 shows the hydraulic unloader.



HYDRAULIC OPERATED UNLOADER

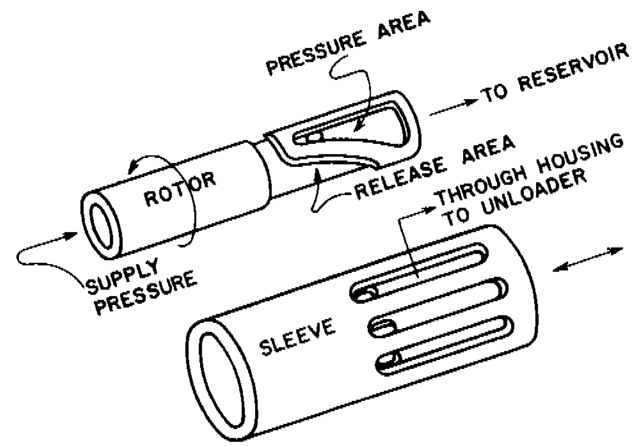
FIG. 4

Oil pressure enters through a check valve and moves the unloader piston and plunger downward after the inlet valve has normally opened. Note that the valve is not driven open but opens normally. The plunger comes to rest on the valve channels and prevents them from closing. When hydraulic pressure is released, the pressure drop across the valve and lifting spring drive the plunger and piston upward. The rate at which this motion takes place is determined by the volume of oil displaced, the size of the orifice, and oil viscosity. When the piston covers the orifice, motion is stopped against the trapped volume of oil. This design effectively prevents the valve from failing when it is allowed to close.[1]

DEVELOPMENT OF DISTRIBUTOR

Initial testing was done on a small single stage, single cylinder air compressor. A modified, diesel engine fuel injection pump was used to apply pressure to the unloader. A relief valve in the piping to the unloader limited the pressure on the unloader piston. This pump was timed to the crankshaft to result in pressure being applied at a fixed crank angle. A rack meshing with a gear on the pump plunger was moved with a pneumatic positioner. Rotation of the plunger, with its helical relief, resulted in the release of pressure at a crank angle which would satisfy the demand for delivered gas. The operation of this system is more fully described in reference [1]. Results were excellent and the system was then applied to a two stage air compressor operating at 720 RPM and discharging at 100 psig. After more than 5000 hours of successful operation, consideration was given to a field test on a large engine driven unit compressing natural gas.

At this time it was obvious that the displacement of commercially available fuel injection pumps was insufficient to operate a large number of unloaders. A rotary type distributor was developed which would distribute hydraulic pressure provided by a separate pump. See Fig. 5. The rotor timed to the crankshaft, fits inside a sleeve which can be positioned longitudinally. Oil pressure enters through the hollow rotor to a pressured area on the periphery of the rotor. When this pressure communicates with a port in the sleeve, oil is delivered through an external housing, not illustrated, to the unloader. The sleeve has a multiplicity of ports spaced in accordance with the crankshaft of the compressor. The single rotor thus delivers pressure to the unloaders of a compressor equipped with several double-acting cylinders, pressure being applied sequentially as required. Pressure remains on the unloaders until the rotor rotates to a point where the port in the sleeve communicates with a release area on the rotor. Oil then escapes back to the reservoir. The actual crank angle at which release occurs depends upon the longitudinal position of the ports in the sleeve with respect to the helix on the rotor. As the sleeve is positioned from left to right, the unloading time is increased and capacity decreased.



HYDRAULIC DISTRIBUTOR
FIG.5

Following some minor problems and design improvements the first unit with infinitely variable control was sold in 1964.[2] The primary purpose for capacity variation in this case was to hold the engine at full torque as suction pressure changed while discharge pressure remained constant. Fig. 6 compares "infinite-step control" versus step control. Note that the engine is loaded to 100% load with infinite-step control at all times. A step control system would result in underloading at all but a few points. As suction pressure decreased, the load or torque on the engine increased. Fuel manifold pressure was used as a measure of engine load. When a predetermined fuel pressure, equivalent to 100% load, tended to be exceeded, a pneumatic signal was delivered to the positioner which moved the sleeve in the hydraulic distributor. Positioning of the sleeve changed compressor capacity until engine load was stabilized at 100%. See Fig. 7.

Any number of control schemes are possible as long as a variation in compressor capacity will satisfy the basic requirement. The prime control may be flow, intake or discharge pressure, constant torque with variable intake and discharge pressure, or temperature control in a refrigeration process.

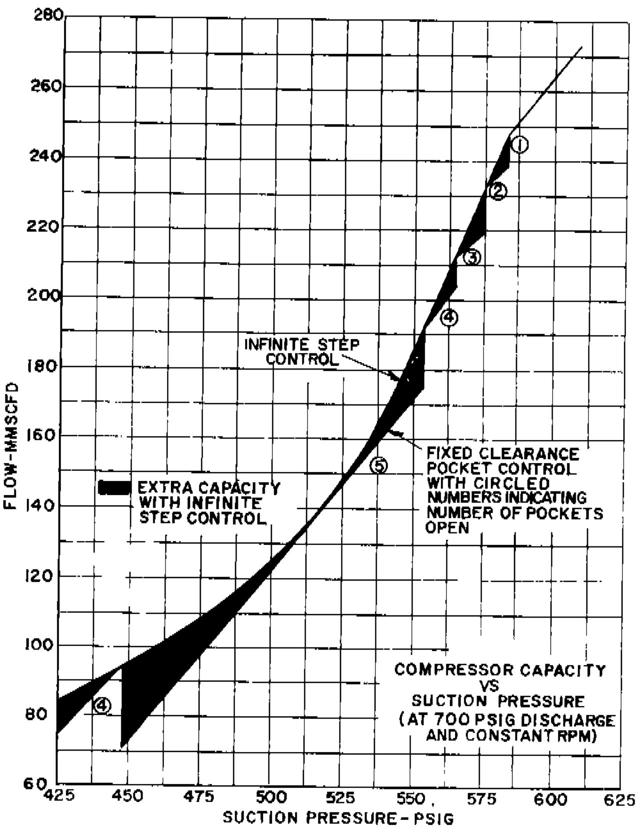
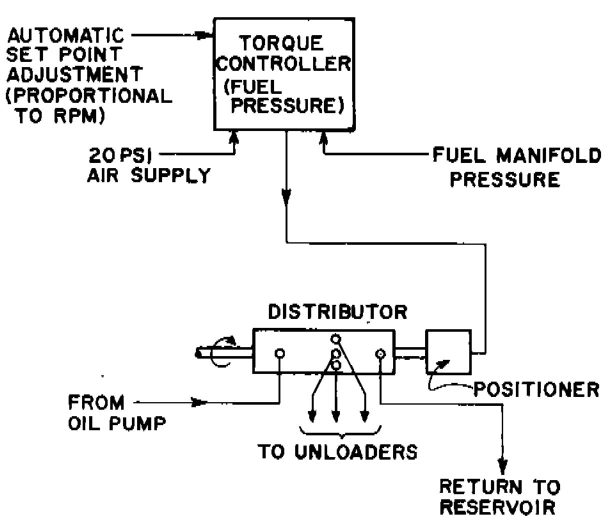


FIG.6



SCHEMATIC CONTROL DIAGRAM

FIG.7

DESIGN PROBLEMS

Problems which arose during development were largely with purchased hardware and not with "in-house" manufactured items such as the distributor and unloader.

A decision which had to be made in the beginning was with respect to the size of the unloader piston and hydraulic pressure. A large piston and low hydraulic pressure would mean relatively large displacements, a large pump, and large diameter tubing from distributor to unloaders to reduce velocity and losses. The final design was to use approximately 3/4" diameter pistons, and 3/8" O.D. hydraulic tubing. High quality, low carbon seamless tubing and a good compression type fitting solved initial problems with these parts. Hydraulic pressure up to 2600 psig has been used successfully. The pneumatic positioner connected directly to the distributor sleeve failed due to high frequency low amplitude pulsations and this was replaced with a pneumatic-hydraulic servo of our own design. In a few cases, excessive wear occurred at the point of contact between the valve channels and unloader legs. This problem was solved by changing the design of the hydraulic piston. A dash pot action was included to slow the plunger down as it approached the channels thus the plunger was now dampened in both directions.

The dynamics of the inlet valve must be carefully considered in this type of system. The valve must open fully without fluttering before the plunger makes contact. Valve flutter can result in unpredictable impact forces between the plunger and channels and cause premature wear or failure.

FIELD TESTING

Field tests were conducted on a large compressor pumping natural gas. Measurements of hydraulic pressure, plunger motion, and cylinder bore pressure, versus time were taken. A motion transducer was attached to the plunger. Piezo-electric pressure transducers measured pressure and all traces were displayed on an oscilloscope. Fig. 8 shows the results of this testing. Starting at the left at zero degrees, the compressor piston is starting its expansion stroke. The pressure expands down to 780 psig inlet pressure and undershoots as the valves open. The distributor applies hydraulic pressure after the valves have opened, and the plunger starts downward. Before reaching the fully downward position, the damping action of the dash pot reduces the velocity of the plunger. Hydraulic pressure peaks at higher than the 2300 psig supply pressure as plunger motion stops. At 180° crank angle, the compressor piston starts its compression stroke but the control system is demanding reduced capacity. At approximately 225°, the distributor starts to release hydraulic pressure. This pressure drops until the lifting spring and back flow forces drive the plunger upward. This action pushes oil back into the unloader tubing and momentarily increases the pressure in this line. Cylinder bore pressure also starts to increase as the valve channels move to their seats. The upward motion of the plunger is damped as the hydraulic piston covers the orifice shown in Fig. 4. With the inlet valves fully closed, compression continues until discharge valves open at 1060 psig.

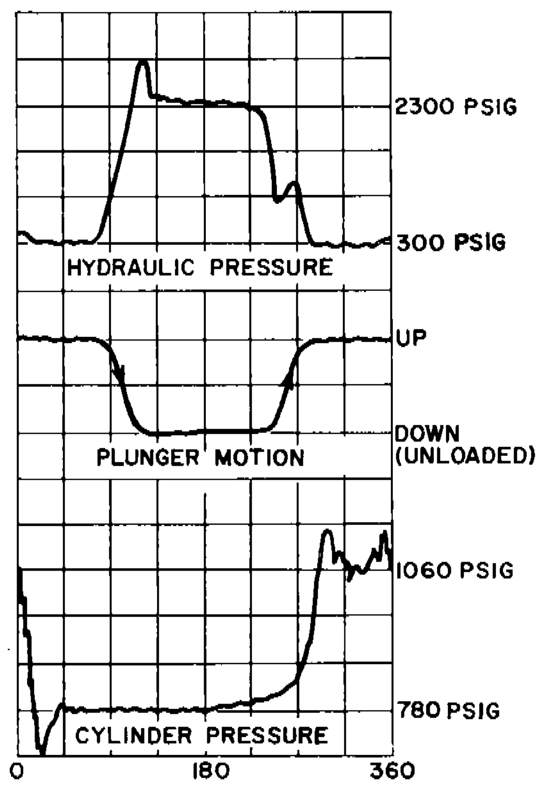


FIG.8 PRESSURE AND MOTION VS CRANK ANGLE

These indicator cards and successful operation without failures confirmed the soundness of the theoretical concepts and design. At this writing a large number of installations are in operation. Several customers have duplicated original units.

ADVANTAGES OF INFINITELY VARIABLE CAPACITY CONTROL

1) Existing hardware is utilized; namely, inlet valves.

2) On double-acting or multi-cylindered units, all ends do equal work resulting in more uniform crank effort.

3) On double-acting or multi-cylindered units, equal volumes of gas are taken from and delivered to manifolds by each cylinder end. This results in simpler pulsation control.

4) When used in conjunction with torque control, full torque can be maintained 100% of the time which is impossible with step control.

5) Changes in flow to the process are gradual and "bumpless".

6) Capacity can be infinitely varied from 100% to 0%.

REFERENCES

1. P. A. Bancel, Ingersoll-Rand Company, U.S. Patent 3,104,801.

2. "Infinite-Step System Provides Close Engine Loading Control" by R. J. Baker and Monroe Martin. The Oil and Gas Journal, June 8, 1964.